● SPECIFICATION●

METHOD AND APPARATUS FOR ROTARY MACHINE VIBRATION CONTROL

Technical Field

[0001]   This invention relates generally to a rotary machine, and more particularly to a method and apparatus for turbine rotor vibration control.

Background

[0002]   A rotary machine such as a turbomachine (e.g., gas turbine, steam turbine, or other turbine system for power generation) is subjected to a variety of forces under operating conditions, particularly when accelerating or decelerating through critical speeds that excite a natural frequency of the system. The magnitudes of the forces and the resulting strains and vibrations on the machine components depend on the machine operating conditions and structural characteristics. Excessive vibration within the machine can result in excessive noise, excessive wear to internal seals and bearing surfaces, increase in secondary leakage in the turbine, and loss of turbine performance and efficiency.

[0003]   The rotors that are employed in turbines are typically long and flexible, are often supported on tilting pads or fixed arc journal bearings, and often operate at speeds in excess of one or more critical speeds. During certain operating conditions, but more particularly during acceleration and deceleration, the rotor may operate at a critical speed (e.g., at a natural frequency) and may experience excessive vibration due either to mass imbalance in the rotor or flex of the rotor.

Summary

[0004]   In one embodiment, a vibration control system for a rotary machine having a rotor includes a sensor disposed within the rotary machine for sensing vibration of the rotor, a vibration damping device disposed within the rotary machine for imparting a reaction force to the rotor, and a controller arranged in operable communication with the sensor and the vibration damping device. The controller is adapted to receive a sensor signal from the sensor and to send a control signal to the vibration damping device for damping the vibration of the rotor.

[0005]   In another embodiment, a rotary machine having a vibration control system for controlling vibrations resulting from an excitation force acting upon the rotary machine includes a rotor bearing housing, at least one bearing supported within the rotor bearing housing, a rotor disposed adjacent the at least one bearing, at least one sensor disposed within the rotary machine to sense vibration of the rotor, at least one vibration damping device disposed between the rotor bearing housing and the at least one bearing to dampen the vibration of the rotor, and a controller arranged in operable communication with the at least one sensor and the at least one vibration damping device. The controller is adapted to receive a sensor signal from the at least one sensor and to send a control signal to the at least one vibration damping device for damping the vibration of the rotor.

[0006]   In a further embodiment, a method for damping the vibration of a rotor in a rotary machine includes sensing the vibration of the rotor, communicating the sensed rotor vibration to a vibration damping device, and damping the vibration of the rotor.

[0007]   In an additional embodiment, a vibration control system for a rotary machine having a rotor includes means for sensing vibration of the rotor, means for calculating a counter-vibration response signal based on the sensed vibration of the rotor, and means for generating a counter-vibration force based on the counter-vibration response signal for damping the vibration of the rotor.

Brief Description of the Drawings

[0008]   Referring now to the figures, which are exemplary embodiments, and wherein like elements are numbered alike:

[0009]   Fig. 1 depicts a generalized perspective view of a turbomachine operative for

implementing the present invention;

[0010]   Fig. 2 depicts a generalized axial cross-sectional view of a rotor bearing housing of the turbomachine of Fig. 1 operative for implementing the present invention;

[0011]   Fig. 3 depicts a generalized perspective view of one type of piezoelectric component and electrical conductors that may be implemented in the present invention;

[0012]   Fig. 4 depicts a generalized perspective view of a stack of piezoelectric components of the type in Fig. 3;

[0013]   Fig. 5 depicts a generalized schematic of a control system for implementing the present invention;

[0014]   Fig. 6 depicts a generalized analytical model of the rotor bearing housing of Fig. 2;

[0015]   Fig. 7 depicts empirical data and analytical curve fitting representative of sensed vibration from a turbomachine of Fig. 1;

[0016]   Figs. 8-10 depict numerical predictions of the analytical models of Figs. 6 and 7 at 2X, 4X and 6X damping; and

[0017]   Fig. 11 depicts a method that implements the present invention.

Detailed Description

[0018]   A detailed description of an embodiment of the present invention is presented herein by way of exemplification and not limitation with reference to Figures 1-11.

Rotor Bearing Housing

[0019]   Figure 1 depicts a generalized perspective view of a turbomachine 100 having a turbine system 200 and rotor bearing housings 300. Referring now to Figure 2, an axial cross-sectional view of rotor bearing housing 300 is shown, which may be constructed of one or more housing sections, such as upper housing 302 and lower housing 304, to facilitate assembly. Radially arranged within rotor bearing housing 300 is a bearing pad support ring 350, bearing pads 400, and a rotor 450, wherein

rotor 450 operates at a rotational frequency of ω (ω =2 Π f, f=rotational frequency in Hertz). Bearing pad support ring 350 and bearing pads 400 may be constructed as individual components as shown, or as an integral unit. Figure 2 depicts five bearing pads 400 supported by bearing pad support ring 350, however, it is understood that the instant invention is not restricted to a particular number of pads, and may be employed where the bearings are either bearing pads, as discussed, or fixed arc journal bearings, roller bearings, or ball bearings, as will be discussed below.

[0020]   In signal communication with rotor 450 is a vibration sensor 500, which may be a proximity sensor (displacement type sensor), velocimeter (velocity type sensor), accelerometer (acceleration type sensor), or any other sensing device that is capable of sensing vibration, which picks up the level of vibration from rotor 450. Sensor 500 is supported by bearing housing 300 and is arranged in signal communication with controller 650 via signal line 600. Controller 650 receives the sensor signal, computes a response signal, as will be discussed below, and sends the response signal via signal line 700 to piezoelectric actuator 750. While signal lines 600, 700 are depicted as single lines, it is understood that these single lines also represent twisted wire pair, ethernet, wireless communication, or any other communication means suitable for transmitting the sensor signal. It is also understood that the devices connected at either end of signal lines 600, 700 are capable of sending and receiving associated signals.

[0021]   The construction of piezoelectric actuator 750 is depicted in Figures 3 and 4, and will be discussed below. Piezoelectric actuator 750 is arranged in operable communication with bearing pad 400 via bearing pad support pin 800. Support pin 800 is arranged to provide a force to bearing pad 400, and therefore to rotor 450, that is directed perpendicular, or trans-axial, to the rotor axis. Support pin 800 is shown to be supporting a single bearing pad 400, however, other individual pads may be supported on separate support pins 800 and piezoelectric actuators 750, or the entire bearing may be supported on a single support pin 800 and piezoelectric actuator 750. The latter arrangement could also be used for fixed arc journal bearings, roller bearings, or ball bearings.

Piezoelectric Actuator

[0022] Figures 3 and 4 depict a generalized perspective view of one type of piezoelectric component and electrical conductors, and a generalized perspective view of a stack of piezoelectric components and electrical conductors, respectively. Referring to Figure 3, a piezoelectric component 850 includes a block of piezoelectric material 852, electrically conductive electrodes 854, 855, arranged at opposite surfaces thereof, and electrical leads 856, 857, connected to the electrodes 854, 855. Piezoelectric material 852, and piezoelectric actuator 750, may be made from ceramic material lead-zirconate-titanate (PZT), lead-titanate (PbTiO2), lead-zirconate (PbZrO3), barium-titanate (BaTiO3), or any other material exhibiting a piezoelectric or polarized electrostrictive effect. The noted ceramics do not, in the pure sense, exhibit a piezolectric effect, but rather a polarized electrostrictive effect. To be truly piezoelectric, a material must be formed as a single crystal, and since a ceramic is a multi-crystalline structure, it must first be polarized to align the individual grains to produce the piezoelectric effect. Since the term piezoelectric has become interchangeable with polarized electrostrictive effect in most literature, the term piezoelectric effect will be used herein for simplicity. Once the piezoelectric material has been polled (polling is the process of applying a one time voltage differential between electrically opposite sides of the piezoelectric material to orient the piezoelectric at the crystalline level, thereby establishing an electrical axis 858), an applied voltage along the electrical axis 858 will result in a corresponding strain along the same axis, thereby providing an electrical-to-mechanical energy conversion (i.e., force) in the direction of electrical axis 858. The piezoelectric device 900 depicted in Figure 4 includes a plurality of blocks of piezoelectric material 852 arranged adjacent one another with electrodes 854, 855, arranged at opposite surfaces thereof, and with electrical leads 856, 857, connected to the electrodes 854, 855. The arrangement of Figure 4 is typically referred to as a piezoelectric stack, and is generally employed to magnify the piezoelectric effect for a given applied voltage. The piezoelectric device (piezoelectric stack) 900 is one type of piezoelectric device that may be employed in piezoelectric actuator 750, however, any device exhibiting a piezoelectric effect is equally applicable.

Closed-Loop Feedback Control System

[0023] Referring to Figure 2, vibration amplitudes associated with rotor 450 are sensed

by vibration sensor 500, which communicates the vibration signal along signal line 600 to controller 650, which calculates a response signal and transmits the response signal via signal line 700 to piezoelectric actuator 750, which receives an input voltage from the response signal and produces a corresponding output deflection to move bearing pad support pin 800 and bearing pad 400 in a motion counter to the vibratory motion of rotor 450, thereby damping the rotor vibration and reducing the magnitude of the vibration signal sent by vibration sensor 500. In the analytical model presented and discussed below, a single vibration mode system is used as an example, however, the controller 650 can deal with multiple vibration modes in practice. The counter vibrations of piezoelectric actuator 750 are preferably at the same frequencies of vibration of rotor 450, however, the amplitudes and phases of the counter vibrations are determined by controller 650 and an imbedded digital signal processor, discussed below in reference to Figure 5, such that the total vibrations at the sensor 500 are minimized.

[0024]   Referring now to Figure 5, control system 1000 is shown to include turbomachine 100, vibration sensor 500, controller 650, piezoelectric actuator 750, and the communicative relationships therebetween. Controller 650 is shown having a signal amplifier 652, an analog to digital (A/D) converter 654, a digital signal processor (DSP) 656, a control algorithm 658, a digital to analog converter (D/A) 660, and a piezoelectric amplifier/conditioner 662. Signal amplifier 652 conditions the sensor signal by converting it into a stable voltage signal for stable transmission and noise rejection. The conditioned signal is then digitized by A/D converter 654 for digital signal processing at DSP 656. Control algorithm 658 analyzes the conditioned signal provided by DSP 656 (system input) and calculates an output response signal that is fed to D/A converter 660 (system output). The system output to input ratio of the control algorithm 658 is typically referred to as a transfer function (H(s)) and is determined by analysis of the system parameters, which is best seen by referring to Figure 6, as will be discussed below. The control algorithm output signal is then converted to an analog signal by D/A converter 660, and then further conditioned by piezoelectric amplifier/conditioner 662 before being fed to piezoelectric actuator 750.

Analytical Model

[0025]   Figure 6 depicts a generalized analytical model 1050 of the rotor bearing housing
of Figure 2, wherein mass (m) 1100 represents the centralized mass of rotor 450,
excitation force (f(t)) 1102 represents the system driving force associated with the
oscillatory motion of rotor 450, displacement variable (x1) 1104 represents the
centralized displacement of rotor 450, spring constant (k1) 1106, and damping
constant (c1) 1108, represent the spring and damping constants of rotor 450,
respectively, displacement variable (x2) 1110 represents the rotor displacement
sensed by vibration sensor 500, spring constant (k2) 1112, and damping constant (c2)
1114, represent the equivalent spring and damping constants of the oil film between
bearing pad 400 and rotor 450, respectively, and displacement variable (x3) 1116
represents the displacement of bearing pad 400. The mass of sensor 500 is
considered negligible in comparison to the mass of rotor 450 and is therefore
omitted. However, it will be appreciated that the omitted mass could be included if
conditions warranted it. Thus, the system equations for the dynamic system of Figure
6, represented in the time-domain where (t) is time, are:

$$mx_1'' + c_1(x_1' - x_2') + k_1(x_1 - x_2) = f(t); \qquad\qquad \text{Equa.-1}$$

$$c_1(x_1' - x_2') + k_1(x_1 - x_2) = k_2(x_2 - x_3) + c_2(x_2' - x_3'); \text{ and} \qquad \text{Equa.-2}$$

$$x_3 = -\alpha x_2 - \beta x_2', \qquad\qquad \text{Equa.-3}$$

where α is a displacement gain constant of piezoelectric actuator 750, and β is a
velocity gain constant of piezoelectric actuator 750. Gain constants α and β are
analytically determined using system control theory and proportional-derivative
feedback control law. Solutions to α and β take into consideration system stability and
control convergence with no oscillations, thereby providing active feedback and
control. Preferably, $\alpha > 1$ and $\beta > 0$. The single prime (') and double prime ('')
notations represent first derivative and second derivative operators, as a function of
time, respectively.

[0026]

          Transforming the above system equations from the time domain to the frequency
domain for ease of signal processing, using known transformation procedures, and
solving for the system transfer function H(s) 658, yields;

$$H(s) = X_1(s)/F(s) \qquad \text{Equa.-4}$$

$$= \quad k_1 + c_1 s + (1 + \alpha + \beta s)(k_2 + c_2 s)$$

$$\overline{\rule{0pt}{0pt}\hspace{7cm}} \qquad \text{Equa.-5}$$

$$ms^2[k_1 + c_1 s + (1 + \alpha + \beta s)(k_2 + c_2 s)] + (1 + \alpha + \beta s)(k_1 + c_1 s)(k_2 + c_2 s)$$

where,

$$s = j\omega, \qquad \text{Equa.-6}$$

$$\omega = 2\pi f, \qquad \text{Equa.-7}$$

$$j = \text{square root } (-1), \text{ and} \qquad \text{Equa.-8}$$

$$f = \text{rotational frequency of rotor 450 (in Hertz).} \qquad \text{Equa.-9}$$

[0027]   Initial values for system variables (m) 1100, (k1) 1106, (c1) 1108, (k2) 1112, (c2) 1114, are established using numerical methods, such as finite element analysis (FEA), and then modified by comparison to empirical data so that the numerical model reflects the fundamentals of the real system. Initial values for gain constants α and β are established using the aforementioned proportional-derivative feedback law, and then modified as discussed above.

[0028]   Armed with modified, or second generation, values for variables (m) 1100, (k1) 1106, (c1) 1108, (k2) 1112, (c2) 1114, α , β , ω and f, the system transfer function H (s) can be calculated, and an output response X1(s) can be determined for a given input F(s). F(s) is provided by the operating conditions of the turbomachine 100 and rotor 450. If the calculated system output response X1(s) differs from the sensed vibration (x2) 1110 as sensed by vibration sensor 500, then controller 650, via control algorithm H(s) 658, adjusts the excitation (x3) 1116 of piezoelectric actuator 750, thereby providing a closed-loop feedback control system for damping undesirable vibrations experienced by rotor 450.

Empirical and Analytical Modeling

[0029]

Figure 7 depicts empirical data 1200 and associated curve fitting 1250, which is representative of sensed vibration from a turbomachine 100 of Figure 1. Using the following empirical parameters from curve fitting 1250;

$c_1 = 11.97 \text{ lb*sec/in}$ Equa.-10

$k_1 = 6078470 \text{ lb/in}$ Equa.-11

System Damping Ratio $\zeta = 3.62\%$, Equa.-12

in combination with the following numerical parameters from numerical simulation

(i.e., analytical model 1050 and control algorithm H(s) 658);

$c_2 = 17100 \text{ lb*sec/in}$ Equa.-13

$k_2 = 13.9e6 \text{ lb/in}$ Equa.-14

$m = 108.7 \text{ lb*s}^2/\text{in}$, Equa.-15

yields a numerical resonance of 1895 rpm. The empirical resonance under the same

conditions, as derived through curve fitting 1250, is 1895.18 rpm, which is a

difference of less than 0.01%, thereby confirming that the analytical model is a

reasonable representation of the real system.

[0030]   As shown in Figure 7, the peak-to-peak resonant amplitude at the 1895/1895.18

rpm resonance is 53 mils, which represents the amplitude of vibration of rotor 450

with no piezoelectric damping.

[0031]   Applying the above empirical and numerical system parameters at varying degrees

of damping, yields peak-to-peak resonant amplitudes, as shown in Figures 8-10, of;

29 mils at 2X damping (Figure 8), 20 mils at 4X damping (Figure 9), and 20 mils at 6X

damping (Figure 10).

[0032]   The degree of damping, system damping, is modified by modifying the value of

damping constant (c2) in Equa.-5, so that (c2) is 2-times (2X), 4-times (4X), or 6-

times (6X), its second generation value. The values of ζ and c2 are related by;

$$\zeta = c_2/2\omega m \qquad \text{Equa.-16}$$

$$= \frac{c_2}{2(2\pi \, \text{RPM}/60)m} \qquad \text{Equa.-17}$$

[0033]   Figures 8-10 depict the empirical 1202, 1204, 1206, and analytical 1252, 1254,

1256, system responses under the 2X, 4X, and 6X damping conditions discussed

above, respectively.

[0034]

By using the analytical model to modify damping constant c2, which represents

the damping effect of piezoelectric actuator 750, an active vibration control system using a piezoelectric actuator 750 can be analyzed. As seen in Figs.8-10 and as discussed above, the use of an active vibration control system as herein described can provide a reduction in rotor vibration at critical frequencies of at least 2:1 (that is, at least a 50% reduction in rotor vibration).

Method

[0035]   Referring to the flowchart of Figure 11, a process 1300 for actively damping the vibration in turbomachine 100 is shown. Process 1300 includes; sensing 1310 the vibration of rotor 450, amplifying 1320 the sensed signal to signal amplifier 652, converting 1330 the analog signal to a digital signal at A/D converter 654, processing 1340 the signal in DSP 656, calculating 1350 a response signal using transfer function H(s) 658, converting 1360 the response signal to analog signal at D/A converter 660, conditioning 1370 the response signal at piezoelectric amplifier/conditioner 662, and converting 1380 the conditioned electrical signal to a mechanical actuation at piezoelectric actuator 750. The flowchart of Figure 11 is presented to outline one sample method of software or logic flow and is not meant to indicate the only method that may be employed.

[0036]   The disclosed embodiment has the advantage of reducing rotor vibration at critical frequencies by at least 50% compared to systems in which the bearing pads are rigidly supported. The disclosed embodiment can also operate under non-damped conditions if the closed-loop feedback control system was to power off, thereby resulting in a fail-safe condition. Additionally, the disclosed embodiment can be turned off during steady-state operation, since under steady-state conditions the rotor typically runs at a speed far-removed from any system critical speed. Furthermore, the disclosed embodiment is compact and retrofittable to existing turbomachines, since turbines typically include vibration monitoring equipment and sensors, but without the disclosed invention.

[0037]   While the invention has been described with reference to an exemplary embodiment, it will be understood by those skilled in the art that various changes may be made and equivalents may be substituted for elements thereof without departing from the scope of the invention. In addition, many modifications may be made to

adapt a particular situation or material to the teachings of the invention without departing from the essential scope thereof. Therefore, it is intended that the invention not be limited to the particular embodiment disclosed as the best mode contemplated for carrying out this invention, but that the invention will include all embodiments falling within the scope of the appended claims.

[0038]   What is claimed is: